HIGH PRESSURE RECIPROCATING PUMP AND CONTROL OF THE SAME

INVENTORS
Robert W. Allington
Scott L. Blakley
Henry L. Walters

BACKGROUND OF THE INVENTION

Field of the Invention

The present invention relates generally to a method and apparatus for configuring and controlling a two-cylinder, reciprocating pump such that it pumps a constant volumetric flow rate at any discharge pressure. Such a pump is useful for applications such as chromatography, which requires accurate, constant flow rates from pumps at high and variable discharge pressure. More particularly the present invention utilizes a constant rotational speed for most of a given cycle, but utilizes pressure control (and slightly varying rotational speed) for a portion of the cycle in which the flow would normally increase if rotational speed is maintained a constant. As a consequence, the pressure control acts to control the rotational speed to a value very close to the previous speed. Another advantage is a largely constant volumetric input flow rate which improves accuracy of low pressure gradient forming. A logic system is incorporated to provide accurate low pressure gradient former switching, corrected for the depressurization of each of the pump cylinders before their refill.

Background Art

Pumps used for liquid chromatography have stringent requirements to deliver constant and accurate flow rates over a range of discharge pressures. Reciprocating (or piston) pumps are usually employed for this purpose for reasons including their relatively fixed displacement.

1

Present-day piston pumps for chromatography are often two-cylinder pumps where the pistons are actuated by cams – usually one cam per piston. The cam profiles are designed so that the sum of the positive (pumping) speeds of the pistons is a constant if the rotational speed is constant. As one piston decelerates near the end of its delivery stroke, the other cylinder has finished filling and the piston accelerates as it begins its stroke toward the cylinder head. The result of the constant sum of piston velocities in this interval is, if both discharge check valves are open, the total flow rate is constant.

When the pump discharge pressure is low (less than about 10 atmospheres, absolute), the flow rate remains constant with constant rotational speed. Liquids are, in fact, not truly incompressible. A measure of a fluid's "compressibility" is the bulk modulus of elasticity, E_v, defined as

$$E_v = -V \frac{dp}{dV}$$

where p is pressure and V is volume. The bulk modulus of elasticity is larger for a liquid than a gas, but is not infinite. According to the above equation, when pressure changes, so does the volume of a liquid. Because of this volume change, and because the components of the pump are non-rigid, the actual travel distance from bottom dead center until the discharge check valve opens becomes non-negligible. The result is, during this portion of the cycle, fluid is being pumped from a single cylinder only, while the shape of the cam profile is increasing or keeping a piston's speed constant. Thus, at a constant rotational speed, the flow rate is less than the constant value required. Early pumps used an accumulator to smooth the pressure and flow rate in time. In present-day pumps, the instantaneous flow deficiency is made up by increasing the rotational speed of the pump by the control system in this region.

To maintain a constant flow rate, the pump discharge pressure should remain constant. Present-day pumps use either a flow measurement or pressure measurement as a process variable within a high-speed control system to maintain constant flow rate. The manipulated variable is the pump's rotational speed. So, during the period in which a piston is traveling toward its head while its discharge check valve is

2

closed, the pump rotational speed must increase.

Control of the speed of the pump is based on pump discharge pressure most of the time, despite the fact that constant speed is required through the majority of the pump's cycle. Control based on pressure is subject to the noise and response time of the sensor. In addition, the flow rate into the pump during the inflow parts of the pump cycle is not well controlled, making it difficult to produce accurate chromatography eluant composition gradients by repetitively and synchronously switching eluant compositions at the pump inlet. This is referenced in the art as "low pressure gradient forming" and has the advantage that it requires only a single high-pressure pump, rather than the usual dual pump.

There is, therefore, a need for a pump to accurately deliver a constant flow rate fluid of controlled (gradient) composition regardless of the discharge pressure.

Summary of the Invention

A purpose of this invention is to provide a method and device capable of producing constant flow rates at a high discharge pressure. It has particular application to chromatography. To accomplish this purpose, in the preferred embodiments, a two-cylinder pump is used, each piston being driven by a separate cam. Both cylinders provide constant flow at high pressure, but are placed less than the usual 180° apart over most of the rotation. For the preferred embodiments, the delivery stroke of each piston is 240° and the refill stroke is 120°. The maximum of a refill stroke of one piston leads the maximum of the delivery stroke of the other piston, and conversely so, by 60°. However, the fluid delivery strokes are 180° apart.

The pumped fluid has a finite bulk modulus of elasticity and the components of the pump are incompletely rigid. Therefore, during a portion of the cycle of each piston, the piston is traveling toward its head, but no fluid is being delivered. This is because the pressure inside the cylinder has not yet reached the value of the pressure in the discharge of the pump, so the discharge check valve is closed. This portion of the cycle of the first piston is called *recompression*. During recompression for one

3

piston, the cam profile for the other piston is shaped such that the other piston is able to deliver the required flow rate. When the pressure inside the cylinder of the first piston reaches the discharge pressure, the discharge check valve for the first piston opens. Because the second piston is still able to supply the full flow rate, if the rotational speed remains constant, the flow rate and discharge pressure will increase. To avoid this unwanted increase in flow rate, the pump's rotational speed must decrease.

Shortly thereafter, the second piston starts to withdraw (retract away from its head), decompressing the high-pressure liquid in the chamber. When this *decompression* portion of the cycle is complete, the second piston's inlet check valve opens and inlets liquid of programmed concentration or composition from a source at (approximately) atmospheric pressure.

Because of the varying flow rate at constant rotational speed, a control system manipulating rotational speed is incorporated into the pump system. During the majority of the pump's cycle, a constant rotational speed will result in a constant flow rate. It is only during the brief period, already described, when the first discharge check valve has just opened, and before a second discharge check valve closes, that the rotational speed must be altered. Therefore, during most of the pump's cycle, the rotational speed is fixed. When the first pistonreaches bottom dead center, the control system switches to pressure control to decrease the rotational speed when appropriate. Discharge pressure control remains in effect until the discharge check valve of the second piston closes, or until the second piston's check valve opens, depending on the particular embodiment of the invention, at which time the pump is returned to fixed speed control.

To effect the required speed decrease, a high-speed control system monitors the pump discharge pressure. This discharge pressure is measured during the portions of the cycle in which the rotational speed is fixed and this pressure is used as a set point for the pressure control system. During the portion of the cycle in which the pressure control system is manipulating the rotational speed, the pressure will be controlled to

4

the pressure set point.

To determine what portion of the cycle in which to apply pressure control, the pressure and the angular position of the cam and motor shaft are monitored continuously. In particular, the angular positions when the pistons are at top dead center and bottom dead center are sensed; and the angular positions where the discharge check valves close (in one embodiment) or the inlet check valves open (in another embodiment) are also sensed and stored into memory, with continuous updating.

A first and a second angular displacement are calculated and stored in memory for each of the pump's pistons. Control is switched from constant angular speed control to constant pressure control at the first angular displacements. Control is returned to constant angular speed control, from constant pressure control, at the second angular displacements.

The two first angular displacements are known to be that of the pistons' bottom dead center locations as detected by a cam or photocell sensor. The opening of a discharge check valve during fixed-speed operation indicates the completion of recompression and can be sensed by a pressure increase. This sudden increase in pressure is correlated to the angular position of the pump shaft, and this *opening* angular-related volume displacement is stored in memory. One such opening volume displacement is stored for each of the two cylinders, comprising a first storage mode. In the same manner, when a discharge check valve closes, due to a piston having come near the end of its stroke, the discharge pressure will experience a decrease as the cam crosses over top high center. This decrease in pressure is correlated to the angular position of the camshaft, and this *closing* angular position may be stored in memory. The pressure disturbances are detected through a high gain factor and are not usually noticeable at the high pressure liquid outlet. One such set of opening and closing angular values are stored for each of the two cylinders.

The volume displacement of recompression, that is, from bottom dead center to the point of discharge valve opening, is calculated from the angular-related volume

displacements explained above.

Since each of the recompression and decompression liquid cycles are very nearly thermodynamically reversible, the decompression volume is very nearly equal to the recompression volume, so the angular position at which decompression ends may be estimated based on that which is known about recompression. In a second embodiment of the invention, control switches back to constant angular speed control from constant pressure control at the end of a cylinder's decompression.

As indicated above, the two respective recompression volumes are stored in memory. These stored amounts are used respectively as decompression points to detect the end of decompression of a respective cylinder. Besides being useful in the second embodiment of this invention, these values are necessary for accurate synchronization of an incorporated low pressure gradient former. By continuing to monitor the pressure, anticipatory, automatic adjustments to these stored displacements are made during operation. An unusual cam design provides for constant fluid inlet velocity as well as constant fluid outlet velocity with respect to constant speed of the camshaft. Constant inlet velocity is not strictly necessary but it decreases the cost and increases the reliability of providing high gradient accuracy.

The novel features believed to be characteristic of this invention, both as to its organization and method of operation together with its further objectives and advantages, will be better understood from the following description considered in connection with the accompanying drawings in which a presently preferred embodiment of the invention is illustrated by way of example. It is to be expressly understood however, that the drawings are for the purpose of illustration and description only and not intended as a definition of the limits of the invention.

Brief Description of the Drawings

Fig. 1 shows a schematic of a pump.

Fig. 2a shows a first control scheme.

Fig. 2b shows a second control scheme.

6

Fig. 3 shows a plot of piston displacement versus angular displacement.

Fig. 4 shows a plot of pressure versus angular displacement.

Fig. 5a shows a pump inlet assembly, including solvent sources, valve tree, and actuation logic controller.

Fig. 5b shows a pump inlet assembly, including solvent sources, a valve having multiple inlets and one outlet, and actuation logic controller.

Fig. 5c shows a pump inlet assembly, including a plurality of two-way valves feeding a manifold.

Fig. 6 shows a cutaway of a cylinder head with inlet and outlet ports.

Fig. 7 shows a logic diagram for low pressure gradient forming.

Fig. 8 shows the cam.

Fig. 9 shows the displacement profile of the cam with cam follower.

Fig. 10 shows the velocity profile of the cam with cam follower.

<u>Best Mode for Carrying Out the Invention</u>

A schematic depiction of a two-cylinder pump **100** is shown in **Fig. 1**. Inside each of the two cylinders **105, 110**, pistons **115, 120** oscillate. The flow entering from the sample passes through inlet check valves **125, 130** that permit the fluid to pass into the cylinder through the inlet line, but disallow the fluid from exiting through the inlet line. Discharge check valves **135** and **140** function to permit the flow of fluid from the pump cylinders **105, 110** but to disallow the fluid from reentering the cylinder from the discharge line.

The velocities of the pistons **115, 120** are dictated by cams **145, 150** that are both driven by a single shaft **155** and followers **146, 151**. A variable speed electric motor **160** provides the motive power to the slower-rotating camshaft **155** through a gearbox **165**.

Several transmitters are shown in **Fig. 1**. The angular position transmitter **170** provides a value for the angular displacement, α, of the camshaft **155**. The angular increment transmitter **175** provides a value of the motor shaft's **163** incremental

angular position $\Delta\theta$. Shaft speed ω is equal to the first temporal derivative of θ, or the update rate of $\Delta\theta$ that takes place in a rate converter **287** (**Fig. 2**). The pump discharge pressure transmitter **180** provides a value for the pump discharge pressure, p, downstream of the discharge check valves **135, 140**. These three values are used by the control system to maintain a constant flow rate. Two other transmitters **137, 142** are optional. They are used to detect the opening of the check valves **135, 140**, respectively.

Schematics of representative control systems are shown in **Figs. 2a** and **2b**. These figures show the controller using random logic to clarify operation. A similarly functional microprocessor could be substituted.

The transmitters **170, 175, 180**, measuring values of angular displacement, α, incremental angular position, $\Delta\theta$, and pump discharge pressure, p, respectively, are shown once again. The pressure signal is filtered, if necessary, in a filter block **205**. The filtered pressure value is then passed on to a storage block **210**. The storage block **210** accepts one or more readings *outside* certain regions. In the first embodiment of this invention (**Fig. 2a**), the regions outside of which pressure readings are accepted by the storage block **210** are $\alpha_0 \leq \alpha \leq \alpha_2$ **340** and $\alpha_4 \leq \alpha \leq \alpha_6$ **360** (as shown in **Fig. 3** and explained below). In a second embodiment (**Fig. 2b**), the regions are $\alpha_0 \leq \alpha \leq \alpha_3$ **380** and $\alpha_4 \leq \alpha \leq \alpha_7$ **385**. If more than one reading is taken, the values may be averaged to smooth out variations due to noise, etc. The resulting value is used as the discharge pressure set point, p_{sp}. The discharge pressure set point proceeds to the pressure controller **215** where it is actively used *inside* the regions $\alpha_0 \leq \alpha \leq \alpha_2$ **340** and $\alpha_4 \leq \alpha \leq \alpha_6$ **360** (**Fig. 3**) in the first embodiment (**Fig. 2a**) as the pressure controller's **215** set point. In the second embodiment (**Fig. 2b**), the pressure set point is used inside the regions $\alpha_0 \leq \alpha \leq \alpha_3$ **380** and $\alpha_4 \leq \alpha \leq \alpha_7$ **385**.

The value of the filtered pressure is also received from the pressure sensor **180** by the pressure controller **215** as its process variable. The error between the process variable, p, and the set point, p_{sp} is calculated by the controller and a rotational speed set point, ω_{sp}, calculated using a control algorithm such as a Proportional Integral

8

Derivative (PID) algorithm. The value of the rotational speed set point as calculated by the pressure controller is only used by the motor speed controller **230** (via the error block **225**) if $\alpha_0 \leq \alpha \leq \alpha_2$ or $\alpha_4 \leq \alpha \leq \alpha_6$ as determined by the logic block **220** in the first embodiment (**Fig. 2a**), and if $\alpha_0 \leq \alpha \leq \alpha_3$ or $\alpha_4 \leq \alpha \leq \alpha_7$ by the logic block **222** in the second embodiment (**Fig. 2b**). The rate converter **287** converts $\Delta\theta$ composition pulses from the incremental angular position sensor **175** on the motor shaft **163** to a signal representing the operating speed ω_{op}. An error block **225** allows the manual entry of the set point speed ω_{sp} **250**.

If it is determined that the angular displacement, α, is outside the closed regions $[\alpha_0, \alpha_2]$ and $[\alpha_4, \alpha_6]$ by the logic block **220** in the first embodiment (**Fig. 2a**) or $[\alpha_0, \alpha_3]$ and $[\alpha_4, \alpha_7]$ by the logic block **222** in the second embodiment (**Fig. 2b**), an angular speed set point, ω_{sp}, **250** is passed into the error block **225**, which subtracts the actual operating speed ω_{op} from ω_{sp} **250**. The result of the error block **225** is a speed error signal, ω_e. The speed error signal, ω_e, is sent to the motor speed controller **230** as shown. The motor speed controller **230** uses the error value of the angular speed, ω_e for maintaining the motor speed, ω_{op}, at the desired value.

A simplified depiction of the pistons' distances from their respective bottom dead centers is shown in **Fig. 3**. Note that the fluid delivery times are twice the refill times. The duration of the line **310** corresponding to refilling of the first cylinder is one half that of the line **320** corresponding to the discharge of this cylinder. The same applies to the refill line **321** and discharge line **311** for the other cylinder.

We will focus just on the piston for which the heavier line, comprising the line **320** corresponding to refill and the line **320** corresponding to discharge, is plotted in **Fig. 3**. Fluid is being drawn into the cylinder through most of the piston's travel away from the cylinder head. This portion of travel is shown as the straight line **310**, where the pump is controlled at a constant angular speed, using the set point, ω_{sp}, emanating from the set point block **250**. Through most of the piston's travel toward its cylinder head, fluid is being expelled to a load such as a chromatograph. This portion of travel is the straight line labeled **320**. At the angular position, α_0 **330**, the

9

piston is at bottom dead center. Its velocity is momentarily zero. Both its inlet and discharge check valves are closed. At bottom dead center, when $\alpha = \alpha_0$ **330**, the motor speed controller's **230** set point switches from a fixed rotational speed value, ω_{sp}, emanating from the set point block **250** to a variable rotational speed value, ω_{spp}, calculated by the constant pressure controller **215**. Along the discharge line **320**, the piston begins its course away from bottom dead center, α_0 **330**, toward α_1 **335**. The process taking place when the camshaft is between the angular positions, α_0 **330** and α_1 **335**, is called "recompression:" during this portion of the stroke, the pressure in the cylinder increases from the suction pressure to the outlet pressure. At α_1 **335**, recompression is completed within the corresponding cylinder and the discharge check valve **135** opens. At this time, both cylinders are delivering liquid. This mode of control (utilizing a different – approximately $\omega_{sp}/2$ – rotational speed set point, ω_{spp}, from the constant pressure controller **215**) continues, in one embodiment through the region **340** between α_0 **330** and α_2 **345**. Once past α_2 **345**, the angular speed set point sent to the error block **225** reverts back to a fixed value **250**, ω_{sp}. In a second embodiment, constant pressure control continues through the region **380** between α_0 **330** and α_3 **346** before returning to constant speed control.

The same cycle is experienced by the other piston, having the plot of its distance from bottom dead center versus its angular displacement depicted by the finer line in **Fig. 3** comprising the line **311** corresponding to discharge and line **321** corresponding to refill. The region in which the rotational speed set point, ω_{spp}, emanates from pressure controller **215** for the first embodiment is the interval **360** between α_4 **355** and α_6 **365**. The constant pressure control region for the second embodiment is the interval **385** between α_4 **355** and α_7 **367**.

To determine the angular displacement values α_2 **345** and α_6 **365** at which to discontinue constant pressure control in the first embodiment, the pressure signal is analyzed in a high-speed data analyzer that checks for a sudden change in the pressure signal **410** shown in **Fig. 4**. Without the constant pressure controller **215** of this invention, the pressure would rise **413, 414** when both pump heads are delivering

fluid, then fall again **420, 430** when one of the cylinders returns to its inlet stroke. With the pressure controller **215**, during the pressure controlled regions **340** and **360** of the cycle, the motor slows down to keep the pressure substantially constant is shown by the dashed lines **411** and **412**.

Valid values of α_2 **345** and α_6 **365** can be determined, for instance, by calculating the first derivative of the signal with respect to time, or by calculating the frequency content of the signal and detecting when the amplitudes of higher frequency values change sharply.

Angular position values α_0 **330** and α_4 **350**, representing the beginning of recompression and, therefore, the points at which control is switched from constant angular speed control to constant pressure control, are detected via a cam or photocell sensor.

Another aspect of this invention is associated with the induction of solvent into the pump. To permit a selection of solvents to be accurately mixed in the pump **100**, a solenoid valve tree assembly **560**, shown in **Fig. 5a**, resides between the solvent source **544** and the pump **100**. The use of a valve tree, in which at most one of several paths can possibly be open at a time to a plurality of solvent sources **544**, simplifies switching functions, as it allows abundant time for the valves to mechanically return after deactivation. When a flow path through the valve tree assembly **560** between solvent source **544** and pump **100** is "open," fluid will not flow if the pump's inlet check valves are closed. Solvents are delivered to pump **100** during the inlet strokes **310, 321** of the pistons **115, 120**. The valve tree assembly **560** includes two three-port valves, which are a first electrically actuated valve **540** and a second electrically actuated valve **550**. The solvent source **544** includes three solvent sources **510, 520, 530**. If neither valve is actuated, the solvent will flow from solvent source A **510**, through the valves **540, 550** to the pump **100**. When the valve **550** is actuated, its port to the valve **540** is closed while its port to solvent source C **530** is open. Only solvent C **530** will flow in this condition. Any time it is required that either solvent A **510** or solvent B **520** be flowing into pump **100**, valve **550** must be

11

deactuated. When valve **540** is deactuated, its port to solvent A **510** is open. In order for solvent B **520** to flow into the pump **100**, the valve **540** must be actuated while the valve **550** must be deactuated. In this way, any of the three solvents may be drawn into pump **100** while all other solvents are disallowed from flowing.

In practice, actuation of each of the valves **540, 550** is carefully timed within actuation logic controller **570** during the intake strokes **310** of the pump's cycle such that each of the three solvents **510, 520, 530** are accurately metered into the pump **100**. This is accomplished by a number of complementary means. First is measuring the time delay of each valve's opening, which happens at an only roughly fixed time after the valve is electrically activated. This delay time is electronically stored and used to correct the activation delay time of the same valve in the following mixing cycle. This is described more fully in U.S. 5,158,675 assigned to Isco, Inc. and incorporated herein by reference.

A first optional approach to valving at the inlet of the pump **100** is shown in **Fig. 5b**. Here, a single valve **580** with a plurality of inlets and a single outlet is used rather than a valve tree assembly **560** as shown in **Fig. 5a**. The arrangement shown in **Fig. 5b** is especially attractive for larger pumps due to the cost of high-flow valves **540, 550** in the valve tree assembly **560**.

A second optional approach to valving at the inlet of the pump **100** is shown in **Fig. 5c**. Here, a plurality of valves **591–594** each having its inlet connected to a solvent **510, 520, 530, 535** and its outlet to a manifold **590** is used rather than a valve tree assembly **560** as shown in **Fig. 5a**. Again, the arrangement shown in **Fig. 5c** is a less expensive option for larger pumps.

Another aspect of this invention is that the inlet stroke of each pump occurs at a time during a period of constant displacement rate and constant rotational speed of the associated cam **145** or **150**. These displacements are those for using needle bearing style, 7/8 inch diameter roller cam followers **146, 151** riding respectively on the cams **145, 150** (**Fig. 1**). The cams **145, 150**, a profile of which is shown in **Fig. 8**, are designed such that the displacement rates are linear using these cam followers. The

12

actual displacement versus angular position is shown in **Fig. 9**. The piston velocity is, therefore, remarkably constant as can be seen in **Fig. 10**. **Fig. 3** shows cam and follower displacements that come to pointed peaks for clarity. It is more practical to use rounded peak segments in actual mechanical systems. This does not affect the accuracy of operation since it is based on reading and comparing pressure and cam profiles on an on-line cam angle to piston (or cam) displacement in such a way that error due to fuzzy or rounded flow rate peaks is cancelled. This cam arrangement combined with the operation circuit shown in **Fig. 2** greatly simplifies and increases the accuracy of the gradient valve switching.

Calculating completion of decompression, the time a pump cylinder starts filling, is necessary for starting the gradient former, if accurate gradients are to be made (**Fig. 2**). Decompression is shown as the region between α_2 **345** and α_3 **346** in **Fig. 3** for one piston. The other piston's decompression region lies between α_6 **365** and α_7 **367**. $\Delta\theta$ is the incremental camshaft position and is detected by $\Delta\theta$ photocell interrupter **175**. The rate at which $\Delta\theta$ changes is proportional to ω_{op}, the motor shaft rotational speed operating point. The rate converter **287** converts $\Delta\theta$ to ω_{op} and transmits it to the speed error calculation block **225** and pressure controller **215**. $\Delta\theta$ is also counted by the two-gated counters **291, 293**, which prevent gradient and gradient cycles from operating during depressurization. During recompression, the gated counter **291** counts up from α_0 **330** (cam determined) to α_1 **335** (pressure determined) to determine repressurization of cylinder 1. The up count is first divided by two in the counter **179** because the delivery stroke is twice as long as the filling stroke. The signal processing may be compensated if necessary to match the cam peak profile, or the converse. Then the gated counter **291** down-counts the number of $\Delta\theta$ increments during the depressurization time of cylinder 1 from α_6 **365** to α_7 **367** the latter of which is a count down to zero, whereupon the counter stops. Angular position, α_0 **330**, is determined from the position of the camshaft **155**. Angular position, α_1 **335**, is determined by a pressure increase **413**. The difference between α_0 **330** and α_1 **335** equals twice the up-count angle from α_6 **365** to α_7 **367**, the decompression angle for

13

cylinder 1. (α_7 367 can be calculated since α_6 365 is determined by the cam and the difference, $\alpha_7 - \alpha_6$, can be computed from the difference, $\alpha_0 - \alpha_1$.) Cylinder 2 has its recompression/decompression angle computed similarly by gated counter 293. The gradient former shown in **Fig. 5** is activated (refill) during operation from α_7 to α_0 for cylinder 1. The control for cylinder 2 is similar, activating the gradient former for α_3 to α_4.

Values of θ may be used for the program gradient operation, but corrected values proportional to actual displacement are better as they are not inaccurate because of small departures of cams from ideality. For example, it is not possible to make a robust cam that produces a point-change in slope as shown in **Fig. 3** at α_2. See **Fig. 10** for comparison of results from a real cam. **Fig. 7** shows the use of a lookup table to derive very accurate displacement values to enable accurate gradient forming.

Referring, now, to **Fig. 7**, the up/down counter 705 converts $\Delta\theta$ pulses from values from the angular increment sensor 175 to values of cam angle θ. The lookup table 710 receives values of cam angle θ from the up/down counter 705 and outputs the corrected instantaneous cylinder displacement, X. However, this displacement is not corrected for the fact that flow does not begin when the piston is at bottom dead center (i.e. α_1 335 is greater than α_0 330). This correction may be done by subtracting the observed values of X at the angular value, α, at which the discharge check valve opens and flow begins, that is, α_3 346 and α_7 367, calculated from α_1 335 and α_5 355 as inputted to Latch 713, which calculates the volume of recompression. By subtracting (in the summation block 715) the value of the displacement when the associated discharge check valve opens from the instantaneous cylinder displacement, the actual volume of fluid discharged from the cylinder, Y, is determined.

The total displacement, D, corrected for decompression is calculated in summation block 717 from knowledge of the total displacement determined from α_2 and α_6 365, inputted to Latch 716; and the displacement associated with recompression.

These corrected volumes Y and D both have the proper zero base and spread.

14

Dividing Y by D and multiplying by 100% in division block **720** produces a value equal to the instantaneous percentage of the volume, Q, delivered during a flow cycle.

The quotient, Q, is presented to a triple decision tree **725, 735, 745** along with externally set and programmed gradient set point values for A%, B%, and C%. In the first decision block **725**, the volume percent, Q, is compared directly with a set point, P, for the influx of solvent A. If the result of decision block **725** is true, all valves will be de-energized to permit the flow of solvent A into the pump **100**.

The value of set point P is subtracted from the value, Q, in summation block **730** to calculate a value, S, which is then compared to the set point, R, for solvent B. If the output of the second decision block **735** is true, the valve **540** will be energized, allowing solvent B to flow into the pump **100**.

The value of S is reduced by the value of R in the summation block **740**. The result is labeled U in **Fig. 7**. The value of U is compared with another set point, T, in the third decision block **745**. If the output of the third decision block **745** is true, the valve **550** will be energized which allows solvent C to enter the pump **100**.

Comparators **725, 735, 745** provide control to a gradient solenoid logic driver (not shown). Obviously, the circuit can be changed for any number of solenoid values and liquids by changing the decision tree. Note that, at most, only one solvent will be flowing at any instant.

If one of the cylinders does not fill completely on the refill stroke, the pressure controller **215** will accelerate the motor. As long as the pressure controller **215** does not pass α_7 **367** and α_6 **365** respectively, the control system will compensate for the cylinders not being completely filled.

Still another aspect of the present invention is depicted in **Fig. 6** with the piston at top dead center. The pump cylinder head **600** (shown in the orientation it will be used) is machined to enhance the mixing of the solvents and solutes. Specifically, the inlet port **610** is located further from the seal **615** than the outlet port **620**. Mixing is enhanced within the space between the cylinder wall **618** and the piston **617** the mixing chamber **630**. In this fashion, the last solution to enter the mixing chamber

15

630 is not the first to exit, as is the case when these ports are located directly across from one another.

Obviously many modifications and variations of the present invention are possible in light of the above teachings. It is, therefore, to be understood that within the scope of the appended claims, the invention may be practiced otherwise than as specifically described.

5